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More Effective Control for Centrifugal Gas Compressors Operating in Parallel

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ABSTRACT

Parallel Operating centrifugal gas compressors are major elements in the chemical industry, in oil production and in natural gas gathering, injection, separation, transmission and LNG production. Large energy savings, reduced repairs and improved automation are possible with more effective solutions for compressor station control.

The authors suggest an improved definition of compressor energy efficiency. They analyze several common approaches to control of parallel compressors and evaluate them in terms of energy efficiency, stability of control, equipment protection and level of automation. An improved system developed by one of the authors will be described theoretically and with an example from field operation.

NOMENCLATURE

b	distance between surge limit line and surge control line, dimensionless
e	deviation of operating point from surge control line, dimensionless
h	specific power consumption
h_a	specific power consumption, absolute units
h_r	specific power consumption, relative units
I/P	current-to-pneumatic transducer
K	slope of surge control line, dimensionless
KW	power, kilowatts
KW_{max}	maximum power, kilowatts
m_1	output of Antisurge Controller
m_2	proportional plus integral output of Antisurge Controller
m_3	open loop output of Antisurge Controller
m_4	output of Performance Controller
m_5	output of Load-Sharing Controller
m_6	proportional plus integral plus derivative output of Load-Sharing Controller
PI	proportional plus integral control algorithm
PID	proportional plus integral plus derivative control algorithm
P_s	pressure in compressor suction, bar
Q_{user}	flow to user, standard cubic meters/minute
Q_{max}	maximum flow, standard cubic meters/minute
R_c	compressor ratio, dimensionless
RPM	revolutions per minute
S	relative distance from surge control line, dimensionless
SCMM	flow rate, standard cubic meters/minute
$SCMM_{max}$	maximum flow, standard cubic meters/minute
T	transmitter
β_i	normalizing coefficents for scaling Criteria S, dimensionless
ΔP_c	pressure differential across compressor, bar
$\Delta P_{o,s}$	pressure differential across flow measuring device in compressor suction, mm H_2O

INTRODUCTION

Centrifugal gas compressors operating in parallel are a major component of the chemical industry, oil production industry, and all phases of the natural gas industry. They are expensive to buy, expensive to repair and expensive to operate. Improving the efficiency and reliability of these compressors is therefore worthy of serious attention.

The efficiency and reliability of these compressors can be severely limited by the control system. Less-than-effective controls, rather than mechanical design, is often the cause of lost efficiency, shutdowns and damage. This paper will analyze several common methods of control, describe improved methods developed by one of the authors, and conclude with an example taken from field operation.

Strategies for load sharing will be examined in particular detail with comparisons, where possible, of energy efficiency, stability and precision-of-control, machine protection and level of automation. An algorithm for an improved load-sharing strategy will be described, and records will be analyzed from a field installations.

In many installations load changes are anticipated for parallel compressor operation. Load changes on a 24 hour, weekly or seasonal cycle. However, in many installations examined by the authors, less-than-adequate provisions have been made in the control system for these load changes. Often load control is manual and very costly in terms of wasted energy and lessened reliability. Whether the load

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control is now manual or automatic, it can, in the authors' opinions, be improved significantly. The user will be well rewarded for any improvements.

In some installations, load changes are not anticipated, steady load being predicted. But only in a perfect world would load demand never change. Only in a perfect world would a station always operate in steady state at the design point of each compressor. In the real world in which we live, many operations that have been designed for steady state at full load have been forced, as experience shows, into load changes. Economic cycles or other events beyond our control may have reduced user demand. Changes in the feedstock or improvements in process design may have changed the compressor load. Moreover, mechanical or control failures in any part of the process may any time forcibly change the load.

A more realistic approach is to design the control system of a compressor station to cope safely and efficiently with substantial load changes. We will show that this approach leads also to more-precise and more-stable pressure control, better machine protection and improved automation.

Many different strategies have been employed to unload and load a compressor station. One common method is to set the speed control manually and run each compressor continuously at full load. Swings in station load are met by recirculation of gas around the compressors. This system is obviously wasteful of energy.

Biasing flow is another common station control method. This method employs a primary or station pressure controller plus separate flow controllers on each compressor. Flow controllers are adjusted to divide the flow between the station compressors in some ratios according to their performance characterstics, e.g. 50 percent, 35 percent and 15 percent. As will be described at a later point, this system requires constant readjustment as operating conditions change and has other shortcomings. A similiar problem will be found with station control strategies that bias compressor speeds.

Another very common method calls for base loading the most efficient compressors either at their maximum flow or at their point of maximum polytropic efficiency. The swings in demand load are then met by modulated control of the less-efficient compressors or by on-off control of these compressors. This system is not energy efficient or sufficiently reliable in our opinion.

We believe that all the above methods can be improved on. In our analysis we shall attempt to show:

a. Simultaneous loading and unloading of the compressors is the most-effective station strategy. Properly controlled, it gives the most energy efficiency, the most precise control, better machine protection and improved automation.
b. Reducing recycle is the most-important component of station energy efficiency. Put another way, station control strategy requires an effective antisurge system for energy efficiency, as well as for machine protection.
c. Regardless of variations in compressor polytropic efficiency, the station should unload so that all the compressors reach their surge control lines simultaneously. We shall propose a loading and unloading algorithm for this strategy which we call the "S Criteria".

AN IMPROVED DEFINITION OF COMPRESSOR EFFICIENCY

First, let us carefully review the definition of compression energy efficiency. The measurement commonly used, polytropic efficiency of the compressor, has major limitations we believe. Polytropic efficiency defines the energy efficiency required to compress the gas between the inlet port and the discharge port of the compressor. However, this is not the energy efficiency of gas delivered to the user, a point of great importance. In normal compressor operation there can be substantial energy losses between the discharge port of the compressor and the discharge header, where the gas is delivered to the user, process, or pipeline. In a compressor system major energy losses can occur through the antisurge valve or through the cooling or quenching, all as a part of normal operation.

SPECIFIC POWER CONSUMPTION

Instead, we shall propose a wider definition of energy efficiency, which we term the Specific Power Consumption "h". It can be defined as follows:

> Specific Power Consumption "h" is the energy input to a compressor system required to maintain the controlled gas parameter and divided by the number of flow units of gas delivered to the discharge header. "h" may be defined in absolute units h_a, or in relative units h_r.

For example, if it takes 500 kw of power to deliver 1,000 standard cubic meters/minute of gas at a compression ratio of 2, then in absolute values $h_a = .5 \frac{kw}{SCMM}$. Units are power divided by flow rate.

In order to compare efficiencies of compressors of different capacities and power consumptions, it is convenient to use a dimensionless measurement for Specific Power Consumption. Units are power divided by maximum power and flow rate divided by maximum flow for a specific compressor.

$$h_r = \frac{KW}{KW_{max}} \div \frac{SCMM}{SCMM_{max}} \qquad (1)$$

Since h increases with energy consumption, note that an increase in h is a decrease in system energy efficiency and vice versa.

Specific Power Consumption, therefore, includes energy costs that must be paid for, but are not measured by polytropic efficiency. The authors believe that Specific Power Consumption gives a more realistic measure of the economic costs of a compression system.

UNLOADING AND LOADING STRATEGIES FOR THE COMPRESSOR STATION

Compressors may be loaded and unloaded by speed changes, changes in guide vane position, throttling or recirculation. We shall use for our examples centrifugal gas compressors with variable speed gas turbine drives and recirculation, controlled for compression ratio.

Analogous results would follow if the controlled parameter was inlet pressure, discharge pressure or flow rate. The reasoning and results would be analagous, too, if the compressors were controlled by adjustable guide vanes or throttles.

The strategy selected for sequencing compressors loading and unloading and the algorithm controlling loading and unloading, significantly influences the station efficiency, as we shall show. Three strategies will be analyzed for specific power consumption under different loading and unloading strategies. We shall compare these loading and unloading strategies for their energy efficiency, machine protection and automation.

Minimum Safe Flow

A centrifugal compressor may be unloaded only to its surge limit line. At that limit, dangerous flow and pressure oscillations begin which can damage or destroy the compressor. To protect the compressor from surge damage, gas is recycled through the antisurge valve (also known as the recycle or bypass valve), before the surge limit is reached. The control line where recycle begins is the surge control line. We may consider this to be the line of minimum safe flow. The calculation of the minimum safe flow is complex. There is a different minimum safe flow for each compressor inlet pressure, head, speed, inlet temperature, gas composition and position of guide vanes. The minimum safe flow will also varies substantially, depending on the effectiveness of the surge control system. The more effective the surge control, the closer the surge control line will be to the surge limit line, and the less the recycle needed.

Calculating Specific Power Consumption

The unloading of compressors within the station may be simultaneous, successive, or mixed. The most efficient strategy can be found by comparatively simple calculations based on an analysis of the performance map of each compressor in the station.

Suppose a station of two identical compressors operating in parallel. Fig. 1 gives the performance curves for two identical gas compressors built by a leading European manufacturer. Assume that a user or process requires that the station maintain a constant compression ratio $R_c = 2.75$. Note that the reasoning which follows would apply equally well if the controlled parameter were discharge pressure, or suction pressure or flow.

Specific Power Consumption in Single-Compressor Operation. The Specific Power Consumption "h_r" plotted as a function of the gas flow rate is shown for single compressor operation in Fig. 2. Points A, B, and C lie on a line of constant compression ratio but varying flow rates. Point A belongs to the maximum performance curve corresponding to the maximum speed and flow. At point B, the design point (the expected operating point of the compressor), the speed and flow rate is somewhat lower. The specific power consumption at B has decreased slightly, which indicates an increase of efficiency. Point C lies on the compressor's surge control line. Between points B and C the specific power consumption has increased slightly and the efficiency decreased slightly.

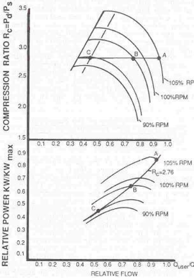

FIG. 1. PERFORMANCE MAP OF A CENTRIFUGAL GAS COMPRESSOR

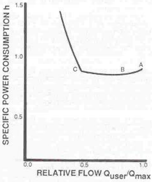

FIG. 2. SPECIFIC POWER CONSUMPTION IN SINGLE-COMPRESSOR OPERATION

At point C, however, efficiency is still higher than at the initial point A.

Point C is the minimum safe flow for this compressor at this speed, with these inlet conditions and with some specific surge control system. From point C, a further decrease of the station flow cannot be safely effected by a further decrease of flow through the compressor. To prevent surge damage, the difference between the station demand flow and the minimum safe flow through the compressor must be recirculated to the suction network. Under such circumstances, the flow to the user decreases. But, the flow through the compressor is constant at C and the compressor power is constant. As a result of this recirculation, the specific power consumption sharply increases to the left of point C. Put in alternate terms:

> The system efficiency of any compressor drops sharply once recirculation begins, and the cost of compressing a flow unit of gas rises sharply.

The effectiveness of the surge control system now becomes critical. The more effective the surge control system is, the narrower the recirculation zone can be. A less effective surge control system will require that the safety margin between the surge control and surge limit lines be increased. An increase of safety margin by only 5 percent will decrease the efficiency at the point C by about 10 percent. Effective surge control systems have been described by the authors in "Improved Surge Control for Centrifugal Compressors". We refer the reader to this article for more details.[1]

In contrast, the change in polytropic efficiency before recirculation begins is not significant at all. The authors have analyzed more than 200 centrifugal and axial compressors controlled by speed, or adjustable guide vanes or throttles. This analysis indicates that the specific power consumption either stays steady as the compressor flow decreases to the surge control line or, in some cases, it increases slightly and then decreases slightly. However, the specific power consumption increases dramatically as soon as recirculation begins.

As further examples, Fig. 3 shows the specific power consumption as a function of flow rate for compressors of four other leading American and European compressor manufacturers. This illustrates our first conclusion for improving the efficiency of compressor operation:

> Increasing the safe operating range without recirculation is more important for compression efficiency than maintaining the compressors operating point at the point of highest polytropic efficiency.

Specific Power Consumption in Multi-Compressor Operation. We will now examine the parallel operation of two or more compressors. Suppose the sample station has two identical compressors. A constant compression ratio must be maintained for this sample station, and the station flow must be adjusted to meet the demand. Note, again, that suction pressure control or discharge pressure control give analagous results.

[1] Chemical Engineering, May 21, 1979

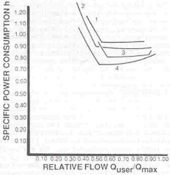

FIG. 3. SPECIFIC POWER CONSUMPTION OF FOUR MAKES OF COMPRESSORS

There are three major alternatives to changing station flow:

a. To unload and load compressors simultaneously
b. To unload and load compressors in sequence, first decreasing the flow of the least efficient compressor and keeping the others at maximum load
c. To combine the simultaneous and sequential unloading strategies.

Specific Power Consumption for Simultaneous Unloading/Loading. Fig. 4 shows the specific power consumption of a simultaneous approach to the surge control lines versus sequential loading/unloading for these two identical compressors in parallel operation. The performance characterstics of these compressors have already been discussed (See Fig. 1). The compression ratio of the station will be maintained during unloading. Again, note that analogous results obtained if the controlled parameter was suction pressure, discharge pressure or flow.

Curve I of Fig. 4 (A, B, C, D, E) shows in relative units the change of specific power consumption using simultaneous unloading of the station. Exactly as with the single compressor, the station efficiency first increases slightly and then decreases slightly between the maximum station capacity (point A) and the minimum safe flow (point B). Both compressors approach their surge control lines at B simultaneously. Between points B and C the station output is adjusted to the lower gas demand by partial recirculation around the compressors.

At point C, compressor No. 1 is shut down. The control system compensates for this flow loss by loading compressor No. 2 to its maximum speed (point D). Then, the station is further unloaded by reducing the speed of compressor No. 2 down to its surge control line (point E). Further decrease in flow to the user is compensated once more by recirculation.

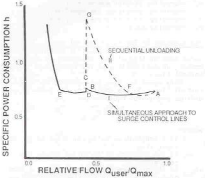

FIG. 4. SPECIFIC POWER CONSUMPTION OF TWO IDENTICAL COMPRESSORS

Between points B and C, the efficiency of the station drops but, after the shutdown of compressor No. 1, it increases at D to the efficiency observed at the initial point A. Efficiency changes insignificantly from D to point E, after which it drastically decreases. Between points A and E, there is no significant change of the station efficiency.

Specific Power Consumption for Sequential Unloading/Unloading. The dotted curve II of Fig. 4 corresponds to sequential unloading/loading. Between points A and F, compressor No. 1 base loaded and kept at its maximum speed. Compressor No. 2 is unloaded down to its surge control line (point F). The efficiency with sequential unloading is less in this segment of curve II than the efficiency under simultaneous unloading.

At point F it is not possible to shut down compressor No. 2 because it it would be impossible to deliver the flow demand. Should compressor No. 2 be shut down at this time, then the flow, the compression ratio and the pressure at the discharge of the station will drop below their required levels. To shut down one compressor without disturbing the required pressure is possible only at point G, where the station efficiency is extremely low. Shutting compressor No. 2 down at G increases the station efficiency to point D. From point D to point E and beyond there is no difference between the curves I and II in efficiency.

Curve II of sequential unloading is nearly always higher in specific energy consumption than curve I of simultaneous unloading between points A and D. This proves, in our opinion, that simultaneous unloading is far more efficient, but only if all compressors reach their surge control lines simultaneously.

For the overwhelming majority of compressor stations equipped with two or more centrifugal compressors, an unloading by simultaneous approach to the surge control lines is the first condition for higher station energy efficiency at all gas demand levels.

Some reflection will show that this conclusion applies to compressors operating in series, as well as in parallel.

Precise Pressure Control

Besides improving energy efficiency, the simultaneous approach to the surge control lines has dynamic control advantages, too. It can give faster and more precise control of station pressure. When the flow demand by the user changes, the pressure differential across the station changes, too. This changes, in turn, the flow through each compressor. If the control system changes the performance of only one compressor at the time, then the overall response time of the system increases. In control terminology, the gain decreases. In order to compensate for the loss of the system gain and to make system response time faster, it is necessary to increase the speed of response of each controller. However, this is difficult to do in some cases because system components such as the valves and transmitters become the limiting factors.

In general the system response can be significantly faster and more precise with simultaneous loading/unloading.

Machine Protection

Comparing the two strategies with regards to machine protection, the simultaneous approach to the surge control line again has the advantage. With simultaneous unloading, the compressors spend a minimum time on their surge control lines as compared to sequential unloading. This is an important feature for damage control, because operation on the surge control line always carries some risk of surge and surge damage.

The compressors also spend less time at their load limit or speed limit, another advantage for machine protection.

There are fewer starts and stops, which is a third advantage.

Automation

Automation is easier with the station having simultaneous loading and unloading. Shutting down a compressor is not indicated until the station flow has dropped to 30 percent. With sequential unloading, shutting down a compressor is indicated at 50 percent of station flow. Therefore, there is less need for operator intervention with simultaneous unloading. The same reasoning applies to loading the station. There will be fewer startups with simultaneous loading. There are other automation advantages possible which will appear in our analysis of advanced station control later in this article.

Unloading/Loading Compressors of Different Sizes

Selecting the unloading/loading sequence for parallel-operated centrifugal compressors may become more complicated when the size, power and efficiency of compressors are different.

Suppose, for example, that one of two parallel working compressors is slightly more efficient (curve 2, Fig. 3) than the other compressor (curve 1, Fig. 3). The more-efficient compressor also has the higher maximum flow and a wider range of operation without recirculation.

The Fig. 5 compares the simultaneous approach to the surge control line (curve II) and sequential unloading (dotted curve I). In the sequential unloading, the less-efficient machine is first unloaded to its surge control line. Between points A (maximum station capacity) and B (the surge control line of the less efficient machine) the station efficiency under the sequential or simultaneous unloading does not show any significant difference for these compressors of slightly different efficiency.

At Point B, efficiency drops drastically under sequential unloading. In contrast under the simultaneous approach to the surge limits, the efficiency remains high until Point C. At C, both compressors must begin recirculating.

Supposing a difference in capacity or efficiency of not more than 7 percent, a simultaneous approach to the surge control lines is the most efficient strategy.

With compressors of significantly different efficiencies (more than 10 percent) mixed unloading or sequential unloading may be more efficient, in our opinion. However, such stations are few in number.

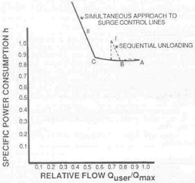

FIG. 5. SPECIFIC POWER CONSUMPTION OF TWO DIFFERENT COMPRESSORS

THE "S CRITERIA" FOR MORE-EFFECTIVE STATION CONTROL

There are many possible algorithms for simultaneous approach to the surge control lines. Obviously, each algorithm depends on measuring the distance from the operating point to the surge control line. Since compressors may differ in capacity and characterstics, the absolute distance of the operating point from the surge control line is not useful. We recommend using a dimensionless number expressing the relative distance from the surge control line. This measurement technique will now be incorporated into a control algorithm.

The algorithm that will be described here we shall call the "S Criteria". It is one possible method of improved control. The S Criteria was developed and patented by one of the authors. We believe this approach to loading/unloading is of value for:

a. controlling a simultaneous approach to the surge limit by all the station compressors,
b. increasing station energy efficiency,
c. improving precision and speed of pressure control,
d. improving machine protection, and
e. improving automation.

The Equation Of The Surge Control Line

To calculate S, the relative distance from the surge control line of any compressor at any given moment, we must begin with the equation of its surge control line.

Many equations have been offered for the surge control line. In the article by the authors[1] we defined one useful equation as:

$$K \Delta P_c + b = \Delta P_{o,s} \tag{2}$$

where:

K is a constant defining the slope of the surge limit line
ΔP_c is the pressure differential across the compressor
$\Delta P_{o,s}$ is the pressure differential across a flow measuring device in the compressor suction,
b is a constant defining the distance in some flow unit between the surge limit and the surge control line.

This equation is valid for compressors with constant geometry, stable gas composition and a surge limit that is linear in the above coordinates. This equation is self compensating for inlet temperature changes. Refer to the article[1] for a detailed analysis of this equation.

Another equation widely used to calculate the surge control line is:

$$f \frac{\Delta P_c}{P_s} + b = \frac{\Delta P_o}{P_s} \tag{3}$$

where:

P_s is compressor suction pressure.

More-complex equations are recommended for compressors with variable guide vanes, for gases with variable composition, or for compressors with irregular surge limit lines. However these simple but widely useful surge control line equations will suffice for this example.

The Equation of the S Criteria

Based on these surge control equations we shall define S as follows:

$$S = \frac{K \Delta P_c + b}{\Delta P_o} \quad \text{or} \quad S = \frac{f \frac{\Delta P_c}{P_s} + b}{\Delta P_o / P_s} \tag{4}$$

If S is less than 1, the operating point is in the safe zone. The lower the value of S, the greater the safe distance from the surge control line.

An S which is equal to 1 corresponds to a compressor operating point located just on the surge control line.

An S higher than 1 corresponds to the operating point having crossed the surge control line and moved toward surge.

There are two alternatives to calculate the deviation "e" of a compressor's operating point from Surge Control Line.

The first is:

$$e_1 = S - 1 \tag{5}$$

The second is:

$$e_2 = f\left(\frac{\Delta P_c}{P_s}\right) + b_1 - \frac{\Delta P_o}{P_s} \tag{6}$$

$$e_2 = \frac{\Delta P_o}{P_s}\left(\frac{f\left(\frac{\Delta P_c}{P_s}\right) + b_1}{\Delta P_o / P_s} - 1\right) \tag{7}$$

or

$$e_2 = \frac{\Delta P_o}{P_s}\,(S - 1), \tag{8}$$

or

$$e_2 = \frac{\Delta P_o}{P_s}\, e_1. \tag{9}$$

Using e_1 in the control algorithm, it is possible to ensure stable operation over the whole range of operating conditions even with a high controller gain. This provides faster, better quality, more precise control.

Controlling a Station by the S Criteria

Fig. 6 shows a station controlled by S Criteria for suction pressure, having two centrifugal gas compressors with gas turbine drives.

The primary or station controller maintains suction pressure. Each compressor has a load-sharing controller that divides the station load, plus an antisurge controller.

The antisurge controllers of compressors No. 1 and No. 2 calculate the criteria S for their respective machines. Each antisurge controller sends its calcuated S value to the calculating module of its companion load-sharing controller.

The calculating module of each load-sharing controller computes the value

$$S' = \beta_1 (S - 1) + \beta_2 \tag{10}$$

This is a universal equation that can be used for any compressor sequencing.

By tuning β_1, it is possible to individually control the rate of change of flow for each compressor between its maximum load and surge control line. This can further improve station efficiency.

If both β_2 are equal then, regardless of the values of β_1, under decreasing flow both compressors No. 1 and No. 2 will reach their surge control lines (S=1) simultaneously. If β_2 are unequal, the compressors will reach their surge control lines sequentially. This permits any sequencing desired.

This value goes to the PID part of the load-sharing controller as its control variable. One single set point for all load-sharing controllers is developed by the primary station controller, which maintains constant station inlet pressure.

The control shown in Fig. 6 is a typical cascade control. The primary controller develops the set points for all the secondary controllers. As is well known from control theory, the stability of cascade control can be achieved only if the secondary controllers are much faster than the primary controller. However, under this controller tuning, when the flow demand changes then the load-sharing controllers would first change the speed of a compressor in a direction opposite to that required to adjust station output to the gas demand. This would increase deviation of suction pressure from its required level and would make the time of the transient process longer. There are two alternatives to avoid this decrease of dynamic precision while using this cascade control scheme.

The first alternative is to install a dynamic filter between the antisurge controller computing S' and the load-sharing controller using S' as its process variable. Such a filter presents a rough model of the controlled object. It permits the primary controller to rapidly change the set point of the load-sharing controllers and correctly change the speed of the compressors to immediately restore pressure to the required level.

The second alternative, patented by one of the authors, is to send feed-forward adaptive signals from the primary controller directly to the control members, temporarily bypassing cascade control. The principal schematic of the author's solution is given on Fig. 7.

The output of the primary station controller not only enters the load sharing controller as its set point, but it also enters the multiplication modules of both the antisurge controller and load-sharing controller. Then, the outputs of these controllers can be defined as:

$$m_1 = m_2 + m_3 + f_2(S) \times m_4 \tag{11}$$

$$m_5 = m_6 + f_1(S) \times m_4 \tag{12}$$

The functions $f_1(S)$ and $f_2(S)$ are shown in Fig. 8.

Making $f_1(S)$ variable permits reducing the gain of the primary or station controller via the load-sharing controller as the operating point of compressor approaches the surge control line. To complete the control system, the primary controller has a symmetrical connection where the gain to the output of the antisurge controller is increased as the operating point approaches the surge control line. Thus, the primary or station controller switches from controlling the speed to controlling the position of the recycle valve while approaching surge. The controllers in this scheme should be set so that the primary controller

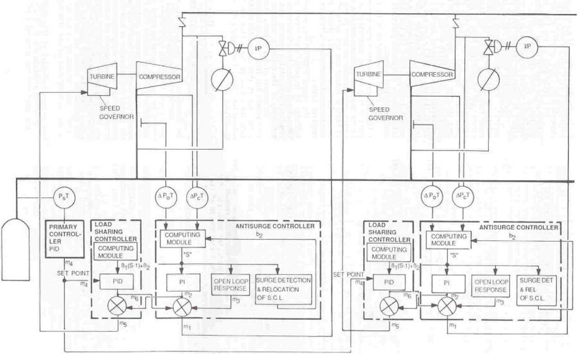

FIG. 6. CONTROLLING COMPRESSORS IN PARALLEL BY S CRITERIA
(SUCTION-PRESSURE CONTROL, LOAD-SHARING CONTROL, ANTISURGE CONTROL)

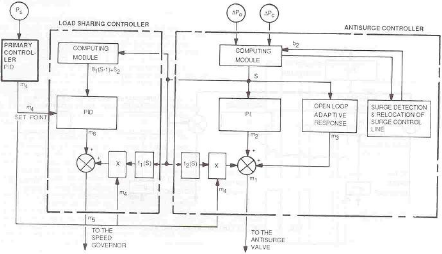

FIG. 7. LOAD SHARING CONTROL BY S CRITERIA

will be much faster than the load-sharing controllers. Then, during changes in gas flow, the station pressure will be controlled first and the load sharing controller will divide the total compressor station flow between the compressors the optimum way.

This scheme achieves energy economy with excellent steady-state precision and dynamic precision. Since the compressors spend a minimum time on their surge control lines or speed limit lines, it adds to machine protection, especially since the number of starts and stops are minimized too.

The use of advanced antisurge controllers adds significantly to the the machine protection. This control is self-compensating and needs no adjustment over a very wide range for changes in inlet pressure, inlet temperature, or flow demand. It, therefore, lends itself to improved automation.

OTHER STATION CONTROL STRATEGIES

Suppose the controlled parameter is discharge pressure and the station consists of two centrifugal gas compressors with gas turbine drives having speed control.

Fig. 9 shows the station controlled by biasing flow. Fig. 10 shows the station controlled by biasing speed. (Biasing the position of the guide vanes or inlet throttle would be analagous control schemes).

In both cases, the settings for each load controller, whether it be controlling through speed or flow, is valid for only one set of inlet conditions.

A glance at the compressor performance map Fig. 1 confirms this. The intersection of the discharge pressure control line and the surge control line for each compressor will correspond to a different speed, or flow rate, or guide vane position whenever inlet conditions change. That is to say, any change in suction pressure, suction flow, temperature, or molecular weight.

Therefore, a simultaneous approach to the surge control line will require retuning of the controllers for optimum load distribution. This is a nearly impossible task for an operator and no easy task for a computer.

A second defect of these commonly used systems is the interaction of the independent surge control loops with the pressure control. The surge controller attempts to decrease discharge pressure and increase inlet pressure, while the pressure control system attempts to maintain discharge pressure. These are incompatible goals and the quality of one or more of the control loops is seriously affected. Oscillation or "hunting" is likely to occur from time to time. Manual control will be called for to stabilize the system.

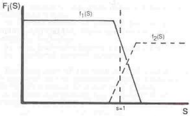

FIG. 8. FUNCTIONS f_1 AND f_2 OF CRITERIA S

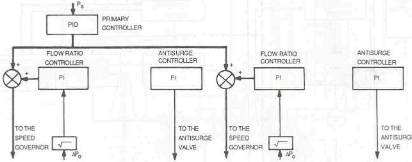

FIG. 9. STATION CONTROL BY BIASING FLOW

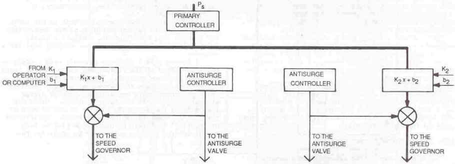

FIG. 10. STATION CONTROL BY BIASING SPEED

A FIELD INSTALLATION

An example of these theories in operation is taken from an off-shore gas compression station equipped with two gas-turbine-driven compressors. The gas supply to this station has both large and fast changes in flow rate. The simultaneously loaded and unloaded compressors use the S Criteria for load sharing to maintain stable suction pressure. The controllers are of the advanced type designed by one of the authors. The controls described here are in continuous operation today.

The station control system was shown in Fig. 6. Fig. 11 shows a strip-chart record of inlet pressure, speed changes of the turbines, and response of the antisurge valves.

The gas supply to the station at the time t_A is low. At t_A, both compressors are operating at low speed (70 percent of the maximum speed of the power turbine for unit No. 1 and 68 percent of the maximum speed for unit No. 2). Both antisurge (recycle) valves are partially open, and the gas supply is starting to increase. The control system simultaneously closes both antisurge (recycle) valves in response. A sharp increase in gas supply momentarily causes inlet pressure to increase at time t_B. The control system rapidly increases the speed of both compressors to restore inlet pressure to the required level. After a slight decrease, the gas supply increases again sharply at time t_C. Again, the control system maintains the inlet pressure by rapidly speeding up both compressors. The control system maintains stable inlet pressure under further sharp fluctuations of the gas supply during the time intervals t_D to t_E and t_F to t_G.

In general, the deviation of the inlet pressure from its required level never exceeded 2 percent. This record shows the precise control, fast response and control stability possible with simultaneous approach to the surge control lines using the S criteria.

Note that the recycle is very low due to:

a. the simultaneous approach to the surge control lines using the S criteria, plus
b. the advanced antisurge control.

A comparision of before and after energy efficiency is not possible, in this case, because this control system is an orginal installation, not a retrofit. Machinery repairs have been minimal, and the need for operator intervention or manual control is negligible.

User reports to the authors indicate that, if considerable load variation exists, an energy savings between 10 percent and 30 percent can be anticipated. Even when the load is stable, significant benefits can be expected from precise pressure control, improved machine protection, and improved automation.

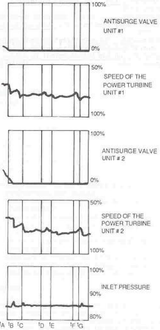

FIG. 11. OPERATION OF A COMPRESSOR STATION, SIMULTANEOUSLY UNLOADING BY CRITERIA S

SOME CONCLUSIONS

Though much attention has been paid to more efficient compressor design, the control systems, which strongly influence operating efficiency and reliability, are often neglected.

Experience shows that we cannot commit the future; that is, we cannot guarantee stable operating conditions, good machine maintenance or trained operators. The authors believe, therefore, that the control system should be designed to insure safe, efficient, automatic operation under all possible operating conditions.

To improve control of parallel (or series) operating centrifugals, there are two key points that are commonly neglected:

a. The loading/unloading strategy, and
b. the surge control strategy.

The extra cost of the best control system is very small compared to the compressor repair costs and lost operating efficiences of a mediocre system.

A serious approach to improved control of centrifugal compressors will be well rewarded.

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